

Thermodynamic analysis of a β type Stirling engine with a displacer driving mechanism by means of a lever

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ABSTRACT

In this study a novel configuration of β type Stirling engine was described and studied from kinematic and thermodynamics points of view. Some aspects of the novel engine were compared to the crank driven and Rhombic-drive engines. By means of nodal analysis, the instantaneous temperature distribution of working fluid, through the heating-cooling passage, conducting the cold space to hot space, was studied. Variation of work generation due to leak of the working fluid was examined and an estimation of the clearance between piston and cylinder was made. By using three different practically possible values of convective heat transfer coefficient, which were 200, 300 and 400 W/m² K, respectively, variation of work generation with working fluid mass was examined. For the same values of convective heat transfer coefficient, the variation of engine power with engine speed was examined. A simple prototype was built and tested with no pressurized ambient air. By applying 260 °C temperature to the hot end and 20 °C temperature to the cold end of displacer cylinder 14.72 Watts shaft power was measured. Results of theoretical study and experimental measurements were presented in diagrams.

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1. Introduction

For the current situation most of the renewable energy investigations concentrates on solar energy. Despite that solar energy is investigated for a large number of purposes, such as electricity generation, water heating or cooling, air conditioning, drying, desalination etc., the prime objective is the electricity generation.

Energy conversion from the solar radiation to electricity is performed by means of photovoltaic solar cells and heat-engines which work with Stirling cycle. For photovoltaic cells up to 24% efficiency [1,2] and for heat-engines up to 30% efficiencies [3] were reported. As variety, too much Stirling engines were built and tested for solar energy conversion. The first application of Stirling engine to solar energy is made by Parker and Malik in 1962. In their conversion system, solar radiation was concentrated and reflected on to the heater of the engine by using a Freshnel lens. The engine used was a piston-displacer machine with Rhombic-drive mechanism [4].

One of the most novel applications of the Stirling cycle is the free piston configuration that initially was designed by W. Beale at Ohio University in the late 1960s [5]. Free piston Stirling engines have no kinematic mechanism coupling the reciprocating elements to each other or to a common rotating shaft. Free piston Stirling engine is

a completely closed system and escape of working fluid from its working space is impossible. Work generation is performed by means of vibration of its body. There are various types of free piston Stirling engines. In solar energy applications β type machines were used and 33% conversion efficiency from light to electricity was obtained [5,6]. This machine usually is connected with a linear alternator to produce electricity.

After 1990, in solar energy applications the low temperature differential (LTD) Stirling engines become fashionable which was invented by White in 1983. A LTD Stirling engine can be run with small temperature difference between the hot and cold ends of the displacer cylinder [7]. It is different from other types of Stirling cycle engines that have a greater temperature difference between the two ends. LTD engines may be of two designs. The first uses single-crank operation where only the power piston is connected to the flywheel, called the Ringbom engine. This type of engine, that has been appearing more frequently, is based on the Ringbom principle. A short, large-diameter displacer rod in a precise-machined fitted guide has been used to replace the displacer connecting rod. The other design is called a kinematic engine, where both the displacer and the power piston are connected to the flywheel. The kinematic engine with a normal 90 degree phase angle is a gamma-configuration engine. Some characteristics of the LTD Stirling engine are: displacer to power piston swept volume ratio is large, diameter of displacer cylinder and displacer is large, displacer is short, effective heat transfer surfaces on both end plates of the displacer cylinder are large, at constant volume heating and cooling

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Nomenclature

A_i	nodal values of heat transfer surface (m^2)	p_{ch}	crank case pressure (Pa)
C_p	specific heat at constant pressure (J/kg K)	R	gas constant (J/kg K)
C_v	specific heat at constant volume (J/kg K)	R_{cr}	radius of crankshaft (m)
D	piston diameter (m)	S_L	length of lever arm connected to displacer rod (m)
E_i	enthalpy flow in and out the nodal volumes (J)	T_i	nodal values of working fluid temperature (K)
e_c	length of cold volume (m)	T_c	cold end temperature of displacer cylinder (K)
e_h	length of hot volume (m)	TH	hot end temperature of displacer cylinder (K)
$\frac{H_p}{2}$	distance between piston top and piston pin (m)	$T_{w,i}$	nodal values of heat transfer surface temperature (K)
H_d	displacer length (m)	t	time (sec)
h_i	nodal values of convective heat transfer coefficient ($\text{W/m}^2 \text{K}$)	Δt	period of time steps (sec)
ℓ_d	length of displacer rod (m)	U_c	length from cylinder top to the fixing pin center (m)
ℓ_m	distance of fixing pin and crank pin (m)	V_i	nodal values of volume (m^3)
ℓ_R	rod length of displacer (m)	$\frac{\pi}{2} - \varphi$	conjunction angle of lever arms (rad)
ℓ_p	length of piston rod (m)	β_p	angle made by piston rod with vertical (rad)
m_t	total mass of working fluid (kg)	β_d	angle made by displacer rod with vertical (rad)
m_i	nodal values of working fluid mass (kg)	γ	angle made by slotted arm of lever with vertical (rad)
Δm_i	variation of nodal mass within a time step (kg)	θ	total quantity of crankshaft rotation (rad)
		δ	clearance between piston and cylinder (m)
		ν	kinematic viscosity (m^2/s)

processes working fluid impinges to the hot and cold surface of the displacer cylinder and a large enough heat transfer coefficient occurs, displacer stroke is small and operating speed is low. The performance of LTD engines built and tested so far did not meet the expectations. Most of them produce a power less than 100 W [8,9]. From author's point of view the most appropriate Stirling engine configuration for solar energy applications is that in which the power cylinder and displacer cylinder are concentrically situated. So that, the displacer cylinder will be situated at the mid center of a dish type solar collector. The sunshine will directly be concentrated on to the hot end of the displacer cylinder. This requirement is well met by β type piston–displacer machines.

With respect to the driving mechanism of displacer, the β type engines may be classified into three groups as crank-driven mechanism, Rhombic-drive mechanism, driving mechanism by means of levers. Cycles performed by all of these driving mechanisms differ from the ideal Stirling cycle due to mechanical reasons. With respect to the degree of diversity, performance parameters of practical cycle decreases. Among the performance parameters, the cyclic work generation is too important because it determines sizes of the engine.

In crank-driven mechanism, during the expansion period, the displacer is stationary about the bottom dead center of its stroke. The working fluid expands into the cold space between piston and displacer. Therefore, the expansion is a polytropic process in spite of the fact that it was a constant temperature process in ideal Stirling cycle. In rhombic-drive mechanism expansion process is performed within the hot space and a better approximation to constant temperature process is obtained. The heating process at constant volume is also fair enough in Rhombic-drive but cooling is not as good as heating. Among the β type Stirling engines, rhombic-drive presents the largest cyclic work generation. For a solar energy module consisting of a dish concentrator and a Stirling engine whose displacer cylinder is situated at the focus of dish, the weight and volume of rhombic-drive is not appropriate.

This study is concerned with the design of a novel arrangement of β type Stirling engine having appropriate weight, volume, external shape and low running temperature. Design process will be conducted by using the nodal analysis presented by Karabulut et al. [10]. Some of the performance parameters of the novel arrangement will be compared to crank-driven and rhombic-drive engines. A prototype will be built and tested.

2. Mechanical arrangement and thermodynamic processes

The mechanical arrangement of novel configuration and relevant nomenclature used in the following text is illustrated in Fig. 1.

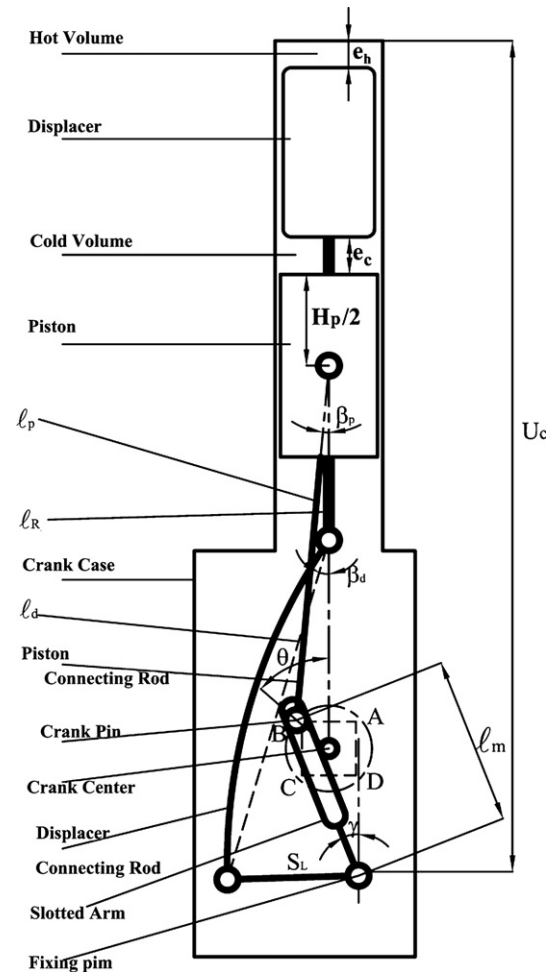


Fig. 1. Schematic illustration of novel arrangement and nomenclature.

The principal difference of the present mechanism from the other mechanisms is that a lever governs the motion of the displacer. On the crankshaft of the present mechanism there is only one pin for the connection of piston rod, displacer rod and lever arm. As seen in Fig. 1, the lever consists of two arms with 70 degree conjunction angle. One of them holds a slot bearing as the other holds a circular bearing. At the corner of lever, a third bearing exists. The lever is mounted to the body of engine through the bearing at the corner by means of a pin. The slot bearing at one arm of the lever is connected to the crank pin. The circular bearing at the other arm of lever is connected to the displacer rod. While the crank pin turns around the crank center, it drives the lever fort and back around the pin connecting the lever to the body. The other arm connected to the displacer rod moves the displacer fort and back.

With respect to Fig. 1, as crank pin moves from A to B, the piston remains about the top dead center of its stroke. By means of performing a downward motion, the displacer expels the working fluid from cold space to the hot space and a heating process at constant volume is performed. As crank pin moves from B to C, piston and displacer moves simultaneously downward and the working fluid expands in hot space by producing work. During this process the cold space increases gradually and some of working fluid fills in the cold space which is undesired. While the crank pin moves from C to D, piston remains about the bottom dead center of its stroke and displacer performs a cooling process at constant volume by means of expelling the working fluid from hot space to the cold space. When the crank pin came to D, the displacer arrives to its top dead center. While the crank pin moves from D to A, displacer remains about the top dead center of its stroke. Piston moves upward and compresses the working fluid within the cold space.

3. Mathematical relations, application procedure and assumptions

The space occupied by working fluid is divided to a number of nodal volumes. First of nodal volumes is the cold space between piston top and displacer bottom and its value is calculated by using kinematical relations. Last of the nodal volumes is the hot space above the displacer top and its value is also calculated by using kinematic relations. Kinematic relations used are,

$$\beta_p = \text{Arcsin}\left(\frac{R_{cr}}{\ell_p} \sin \theta\right), \quad (1)$$

$$\gamma = \text{Arctan}\left(\frac{0.7071 + \sin \theta}{2.5 + \cos \theta}\right), \quad (2)$$

$$\beta_d = \text{Arcsin}\left(\frac{S_L}{\ell_d} \cos(\gamma - \varphi) - \frac{R_{cr}}{\ell_d} 0.7071\right), \quad (3)$$

$$\ell_m = \frac{R_{cr}(0.7071 + \sin \theta)}{\sin \gamma}, \quad (4)$$

$$e_c = \ell_d \cos \beta_d - S_L \sin(\gamma - \varphi) + \ell_R - \ell_m \cos \gamma - \ell_p \cos \beta_p - \frac{H_p}{2}, \quad (5)$$

$$e_h = U_c - \ell_d \cos \beta_d + S_L \sin(\gamma - \varphi) - \ell_R - H_d, \quad (6)$$

Other nodal volumes are in the flow passage between displacer and cylinder and have constant magnitudes. All of the nodal volumes are open systems through which unsteady flow occurs. The relation

between crankshaft angle and time is $\theta = \omega t$. The pressure difference between nodal volumes due to flow friction is disregarded and pressure is calculated by

$$p = \frac{m_t R}{\sum_{i=1}^n V_i / T_i}, \quad (7)$$

For the calculation of temperature variation within a nodal volume between two steps of time, the first law of thermodynamics given for open systems

$$\Delta T = [h_i A_i (T_{w,i} - T_i) \Delta t - \Delta m_i C_v T_i + E_i - p \Delta V_i] / (m_i C_v) \quad (8)$$

is used. In last equation E_i is the enthalpy flow in or out the nodal volume and the approximation

$$E_i = -C_p \frac{T_i + T_{i+1}}{2} \sum_{j=i+1}^n \Delta m_j - C_p \frac{T_{i-1} + T_i}{2} \sum_{j=1}^{i-1} \Delta m_j, \quad (9)$$

may be used. Instantaneous values of mass, m_i , are calculated by using the general state equation of perfect gases. Boundary temperature of flow passage is assumed to be constant with time and varies linearly in flow direction from T_C to T_H . Specific heats are assumed to be constant. For the calculation of leak through the clearance between piston and cylinder,

$$\frac{dm_t}{dt} = \frac{\pi D \delta^3}{12 \nu H_p} (p - p_{ch}) \quad (10)$$

is used. Last equation is the solution of momentum equation for fully developed laminar flow between two parallel plates.

4. Results and discussion

The fixing location of lever, length of lever arms and conjunction angle of arms were determined by means of optimizing the isothermal work generation. Regarding the manufacturing facilities and external dimensions of the engine, the crank radius R_{cr} was chosen as 30 mm. The vertical distance between crank center and lever fixing pin center was determined as $3R_{cr}$. The horizontal distance between crank center and fixing pin center was determined as $R_{cr}/\sqrt{2}$. Optimum conjunction angle of lever arms was determined as 70 degree. Length of lever's horizontal arm (S_L) was determined as 100 mm. With respect to these geometric parameters, the ratio of piston swept volume to displacer swept volume is 0.76. Analysis conducted by Senft [11] using Schmidt thermodynamic model provided the same number for gamma type engines having 90 degree phase angle.

In Fig. 2, kinematic behavior of the novel mechanism and Rhombic-drive engine seen in Fig. 3 are compared. Both engines have the same piston stroke. Maximum and minimum values of cold and hot volume of novel engine are greater than that of Rhombic-drive. Therefore, compression ratio of novel configuration is lower than Rhombic-drive. Maximum and minimum values of cold and hot volumes correspond to different values of crankshaft angle θ . In Rhombic-drive, the minimum value of cold volume appears about 90 degree of crank angle and it remains low enough between 45 and 135 degrees. In novel engine, however, the minimum value of cold volume appears about 50 degree of crank angle and, before and after 50 degree it performs significant variations. This interval of crank angle corresponds to expansion period of working fluid and Rhombic-drive generates higher work by expanding the working fluid in hot space. The interval of crankshaft angle from 135 degree to 225 degree corresponds to cooling process at constant volume. In this process the novel configuration presents a better performance by means of minimizing the hot

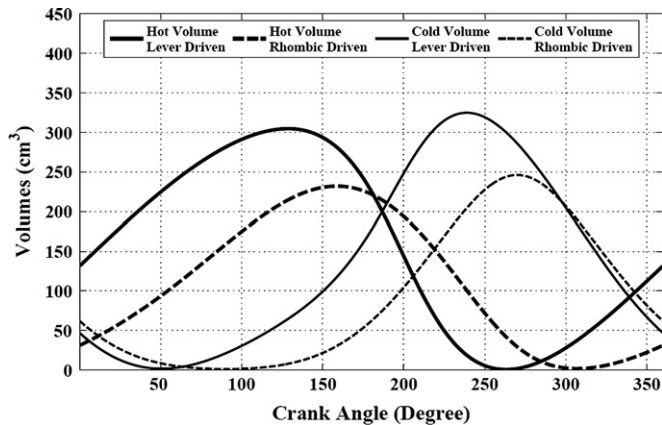


Fig. 2. Comparison of kinematic behavior of novel engine with Rhombic-Drive.

volume and maximizing the cold volume without changing the total volume.

As crank angle varies from 225 to 315 degrees, the working fluid experiences a compression process. We expect that the hot volume will be low enough and constant, as the cold volume is large and decreasing. When compression process is started at 225 degree, cold and hot spaces of Rhombic-drive are equivalent and hot space is decreasing. Since some of working fluid is compressed within the hot volume, Rhombic-drive does not meet our expectation. On the other hand, at 225 degree, the hot volume of the novel engine is low

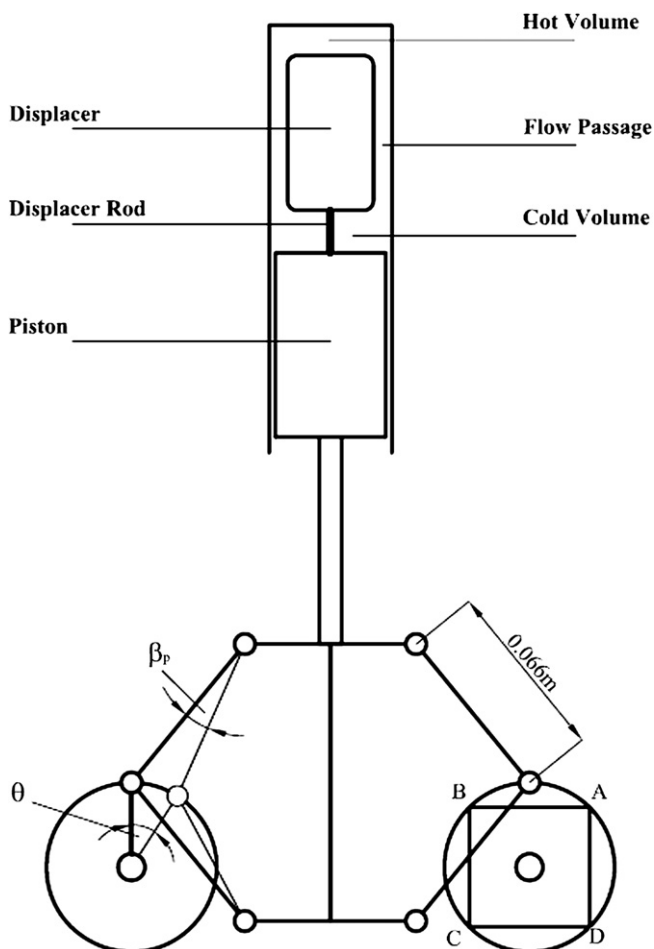


Fig. 3. Schematic illustration of Rhombic-Drive engine.

enough and between 225 and 315 degrees its variation is insignificant. The cold volume is large enough and decreasing. Therefore, novel engine provides a better compression process by compressing the working fluid within cold volume. Through the remaining interval of crankshaft angle, 315–45 degrees, both engines perform a successful heating process. The principal superiority of Rhombic-drive engine rises from its higher compression ratio.

Unless a defined contrary situation, results discussed through the rest of this section were obtained for 350 K cold source temperature, 1000 K hot source temperature, 60 mm piston stroke, 70 mm piston diameter, 1200 rpm speed of engine, 100 cm³ flow passage volume, 1300 cm² heat transfer surface in flow passage and 0.36 gr air mass as working fluid.

In Fig. 4, p–V diagrams of crank driven, Rhombic-drive and novel engine obtained by isothermal analysis under the condition of equal working fluid mass are compared. In crank driven engine the stroke of displacer and piston are equal to each other and the phase angle is 90 degree. In rhombic-drive engine, driving mechanism is a reel rhombus having 66 mm lateral length. Among them, Rhombic-drive engine produces the highest work as 26.31 J. The novel engine and crank driven engine produce 21.56 and 17.66 J works. As seen, right end of p–V diagram of crank driven engine is less blunt than that of novel and Rhombic-drive engine. This is due to the cooling process of crank driven engine, displacer cannot expel all of the working fluid from the hot space to the cold space. The lowest dead volume is seen in Rhombic-drive as the highest is seen in novel engine. Despite that the novel engine has the lowest compression ratio, its work generation is higher than crank driven engine. This is due to the better heating and cooling processes at constant volume. The top limit of novel engine's p–V diagram is lower than others. As long as the work generation is reasonable, the lower top limit provides an advantage by means of reducing the escape of working fluid. In practice, the working fluid is charged to crank case. The working fluid mass in the working space of an engine is assigned by the charge pressure and not constant. Therefore, comparison of engines from the work generation point of view under the condition of same working fluid mass is not a practical evaluation. A comparison of the work generation under the condition of same charge pressure is a better examination to determine the properties of different type engines.

Fig. 5 illustrates a comparison of p–V diagrams of crank driven, Rhombic-drive and novel engines under the condition of same charge pressure. Between piston and cylinder of engines 0.03 mm clearance is assumed. Variation of working fluid mass in the working space is calculated by means of Eq. (7). If 1 bar charge pressure is assumed, the novel engine provides 10.87 J work. Crank driven and Rhombic-drive engines provide 8.46 and 8.47 J works, respectively. This comparison indicates that in practice, the novel engine has advantage. This advantage is caused by the upper limit of fluid pressure is lower than the other engines. Hence, leak of working fluid is less than that of other engines.

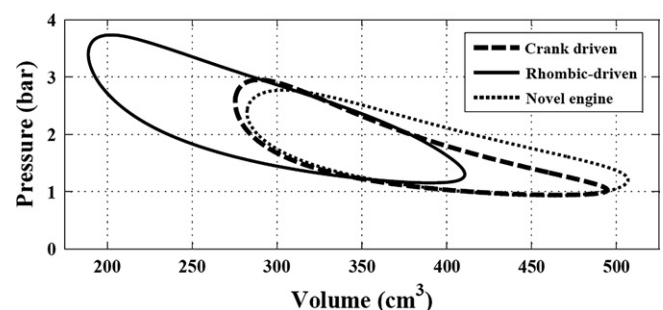


Fig. 4. Comparison of p–V diagrams of crank driven, Rhombic-Driven and novel engine.

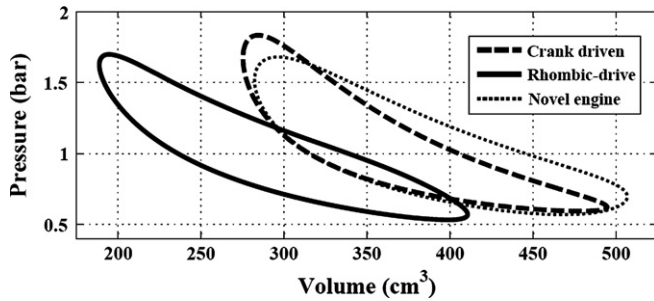


Fig. 5. Comparison of p–V diagrams of crank driven, Rhombic-drive and novel engine in same charge pressure (1 bar).

Fig. 6 illustrates temperature variation of the working fluid within the flow passage in heating and cooling processes. The temperature variation of solid component of the flow passage is assumed to be linear and illustrated on the same diagram. Data illustrated in Fig. 6 are obtained for 1 m^2 heat transfer surface and $300 \text{ W/m}^2 \text{ K}$ heat transfer coefficient in flow passage. Throughout the flow from hot volume to the cold volume, as the fluid temperature was slightly higher than solid temperature, it becomes lower than solid temperature in reverse flow. Towards the finishing ends of flow passage, in both flows an oscillation in temperature profile occurs. The oscillation is caused by the increase of temperature difference between solid and fluid in flow direction. In both flows of fluid, initially the temperature difference between fluid and solid is zero. When fluid advances in flow direction, the temperature difference between solid and fluid increases. After a certain point in flow passage, the term taking into account the heat convection between fluid and solid becomes the dominant term and the mode of equation used for temperature calculation changes. Up to the point mentioned, its solution is an ordinary asymptotic curve. After this point, it becomes an oscillating curve. If the heat transfer surface is reduced more, the change of mode of equation occurs earlier. This oscillation of fluid temperature is not the natural situation. The real temperature of working fluid at any cross section of the passage is the mean of a maximum and a minimum being there. Increase of node number does not generate any difference in oscillation behavior of the fluid temperature. This was confirmed by increasing node number from 50 to 100 as seen in Fig. 6. As long as the other conditions are the same, work and other thermodynamic quantities do not change with node number.

Fig. 7 illustrates comparison of p–V diagrams of novel engine obtained by isothermal and nodal analysis conducted for four different values of convective heat transfer coefficient. As the result,

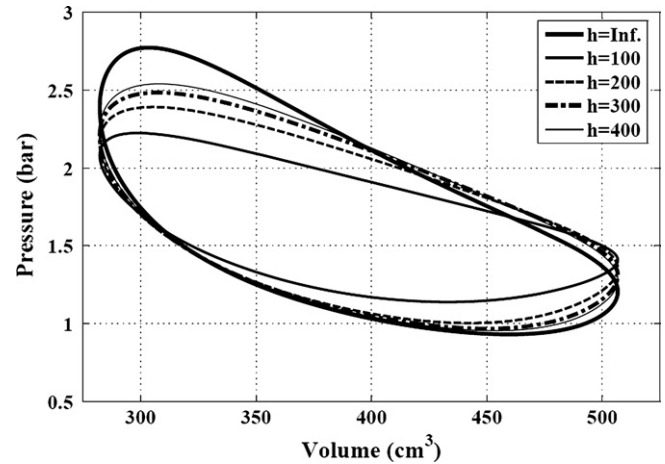


Fig. 7. Comparison of p–V diagrams obtained for different convective heat transfer coefficient.

13.26, 17.97, 19.43 and 20.05 J works are obtained, respectively, for 100, 200, 300 and 400 $\text{W/m}^2 \text{ K}$ convective heat transfer coefficients. Isothermal analysis gives a 21.56 J work which corresponds to an infinite heat transfer coefficient or heat transfer surface. The convective heat transfer coefficients used in this analysis are within the practical limits.

Fig. 8 illustrates effect of the escape of working fluid from working space of the engine. To obtain data used in Fig. 8, the charge pressure used was 2 bars. When the total mass of working fluid in the working space was allowed to vary with respect to Eq. (7), the nodal analysis generates some tedious difficulties. Therefore, data illustrated in Fig. 8 are obtained by using an isothermal analysis modified with Eq. (7). In modified form of isothermal analysis the working fluid mass varies during the cycle with respect to difference of pressures in and out the working space. Between piston and cylinder no sealing, lubrication and roughness were considered. The upper two diagrams seen in Fig. 8 are m–V as the lower two are p–V. When the pressure in working space of the engine becomes greater than charge pressure in the crank case, the leak occurs from working space to the crank case. When the pressure in the working space of the engine became lower than the charge pressure in the crank case, then, the leak occurs from crank case to the working space.

If the instantaneous mass of fluid in the working space is plotted versus the instantaneous total volume occupied by working fluid,

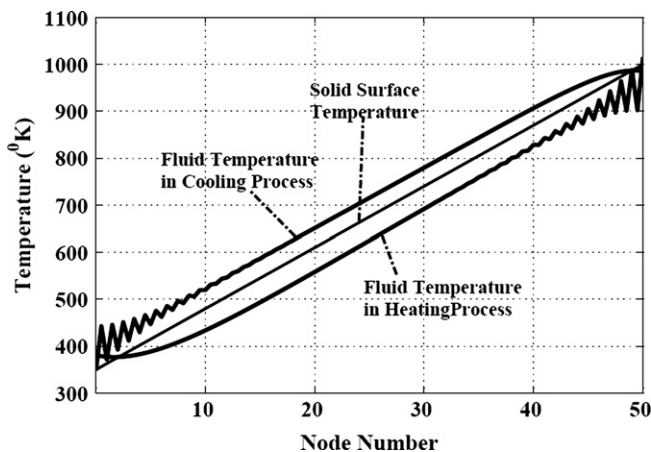


Fig. 6. Variation of fluid temperature through the flow passage.

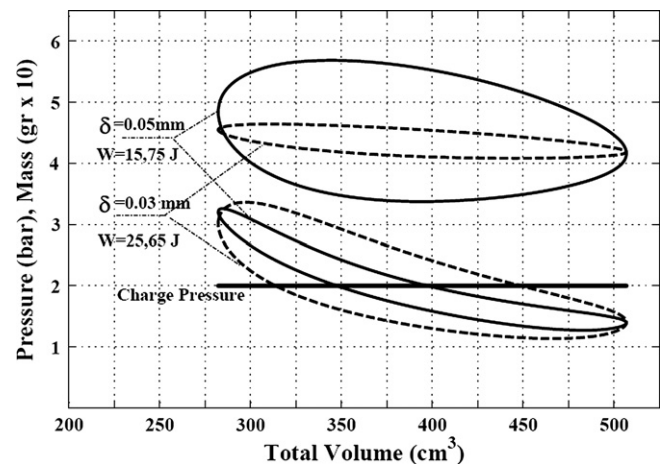


Fig. 8. Effect of working fluid mass variation on the p–V diagram during the cycle.

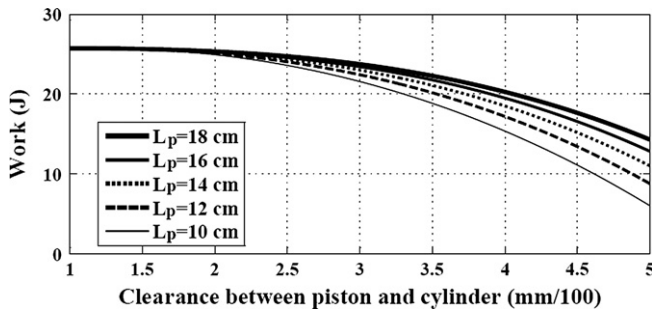


Fig. 9. Variation of work with clearance for several piston lengths.

a closed m–V curve is obtained as seen in Fig. 8. Increase of clearance between piston and cylinder causes the m–V diagram to enlarge. The lowest limit of m–V diagram is a straight line having zero inclination. As seen in Fig. 8, increase of clearance from 0.03 to 0.05 causes approximately 30% decrease in work. Therefore, the appropriate value of clearance is about 0.03 mm.

In Fig. 9, variation of work with clearance is illustrated for several values of piston length where work was calculated by isothermal analysis using 2 bars charge pressure. A larger piston length causes a lesser leak and consequently higher work is obtained. For a ringless piston the appropriate value of clearance is about 0.03 mm. Experimental investigations conducted by Çınar and Karabulut [12] justifies this approach.

Fig. 10 illustrates variation of work with working fluid mass at 200, 300 and 400 W/m² K heat transfer coefficients. For an engine working with air, up and down limits of heat transfer coefficient are estimated as 200 and 400 W/m² K, respectively. Corresponding to 200, 300 and 400 W/m² K heat transfer coefficients 28.8, 43.2 and 57.6 J maximum works are obtained at 1.15, 1.7 and 2.3 gr fluid mass, respectively. In this examination again 1300 cm² heat transfer area and 1200 rpm engine speeds are used. At higher values of working fluid mass, work generation becomes lower. This is caused by the temperature limits of the cycle where up and down limits of the working fluid temperature approach to each other.

Fig. 11 shows variation of engine power with speed. Data used in Fig. 11 are obtained for 200, 300 and 400 W/m² K heat transfer coefficients. For 200, 300 and 400 W/m² K heat transfer coefficients, 1.15, 1.7 and 2.3 gr working fluid masses are used, respectively. Maximum powers are obtained about 1200 rpm, which is a design condition. For 200, 300 and 400 W/m² K heat transfer coefficients maximum powers obtained are 576, 863 and 1152 W, respectively. Below 1200 rpm the engine provides lower power due to lower revolution number per second. Above 1200 rpm, work produced by the engine per cycle is lower than 1200 rpm and due to that, the difference between up and down limits of working fluid temperature becomes lower.

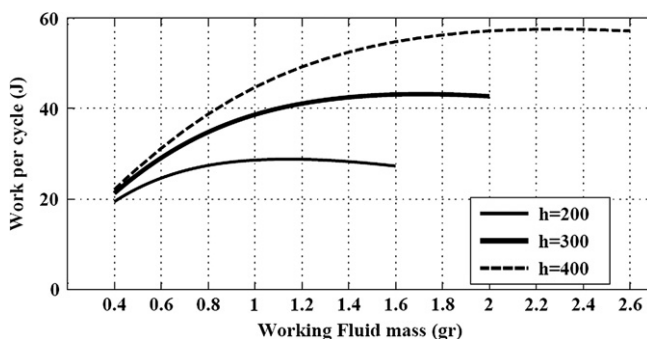


Fig. 10. Variation of work with working fluid mass for heat transfer coefficients 200, 300 and 400 W/m² K.

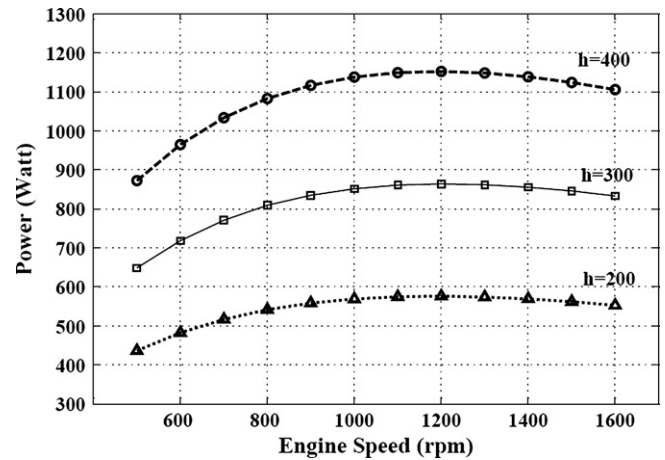


Fig. 11. Variation of engine power with speed.

Fig. 12 shows experimental results obtained from prototype. Working fluid has no pressurized ambient air. The hot end of the displacer cylinder was heated by a LPG flame. To cool the cold end of the displacer cylinder the tap water was circulated in water jacket of the engine. The end temperatures of the displacer cylinder were measured by an infrared thermometer. When the engine started to run, the temperature of hot and cold ends were measured as 98 and 20 °C, respectively. Increase of the hot end temperature continued up to 260 °C and then became fixed. By measuring the speed and moment of the engine, data used in Fig. 12 were obtained. The maximum shaft power was measured at 209 rpm as 14.72 W. This shaft power corresponds to 231 cc piston swept volume and 304 cc displacer swept volume. Under similar working conditions, LTD engines built by Kongtragool and Wongwises [13] provided a shaft power less than 10 W. The swept volume of piston and displacer of LTD engine built by Kongtragool and Wongwises were 894 cc and 6394 cc, respectively.

In Fig. 13, the theoretical results were compared to experimental results. By means of imposing the test conditions to the analysis program, theoretical data used in Fig. 13 were obtained. At low speeds of the engine, the difference between experimental and theoretical values of cyclic work is very small and confirms validity of this analysis. At high speeds of the engine, the difference becomes large. As the speed of the engine increases, inherently, the cyclic work decreases. This is caused by several factors such as increase of mechanical friction and viscous flow losses, decrease of heat transfer between working fluid and solid parts of the engine,

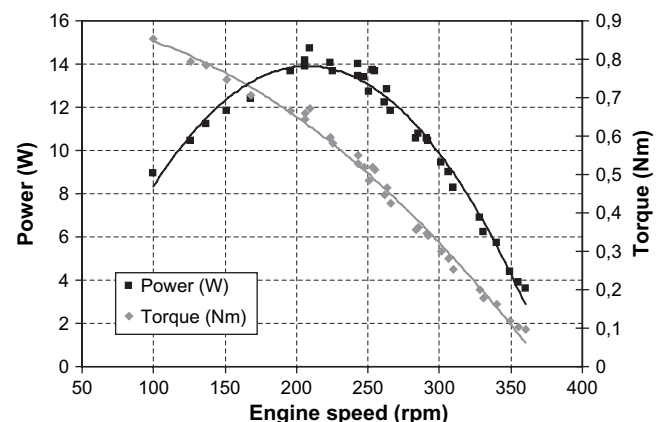


Fig. 12. Experimental results obtained with no pressurized air testing.

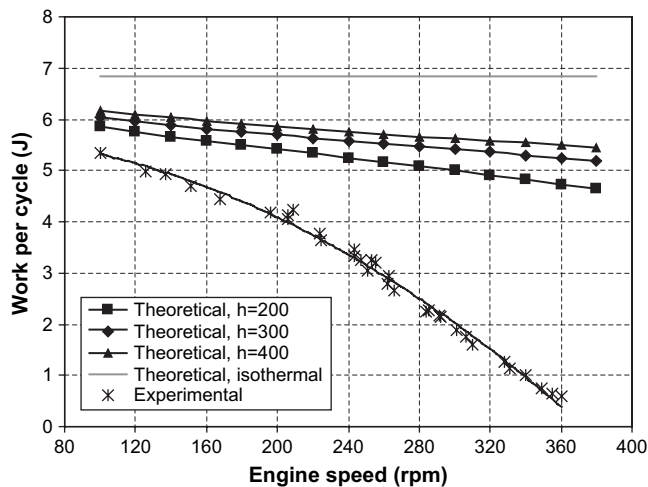


Fig. 13. Comparison of theoretical and experimental results obtained with no pressurized air testing.

increase of dissipative energy due to vibration of the engine, change of the circulation rate of working fluid between cold and hot volumes etc. The mathematical model used in this analysis does not take these factors into account except the heat transfer between the working fluid and solid parts of the engine. Therefore, as the speed of the engine increases, the experimental work decreases steeper than the theoretical work as seen in Fig. 13. At low speeds of the engine, the smallness of experimental and theoretical work's difference is an indication of good sealing between piston and cylinder, where, a piston ring is used that is made of brass. Addition of piston ring to the engine increases the experimental work from 13 W to 14.72 W.

5. Conclusions

Comparison of novel engine with crank driven and Rhombic-drive engines indicates that the compression ratio of novel engine is lowest among the three engines. Comparison of novel engine with crank driven and rhombic-drive engines with respect to equal

amount of working fluid mass indicates that the Rhombic-drive engine has advantage. If comparison is made with respect to the same amount of charge pressure, novel engine becomes advantageous. Clearance between piston and cylinder was estimated as 0.03 mm for ringless pistons ranging from 100 mm to 180 mm length. Temperature distribution calculated by the first law of thermodynamic gives an oscillating temperature distribution along the connecting passage. For an engine having 1300 cm² heat transfer surface, 1200 rpm engine speed, 70 mm piston diameter and 60 mm piston stroke, 28.8, 43.2 and 57.6 J works were calculated corresponding to 200, 300 and 400 W/m² K heat transfer coefficients. Maximum powers corresponding to 200, 300 and 400 W/m² K heat transfer coefficients were calculated as 576, 863 and 1152 W, respectively. The Prototype of the engine provided 14.72 W shaft power at 260 °C hot end and 20 °C cold end temperatures at no pressurized ambient air testing. Experimental data obtained at low speeds of the engine confirm validity of the theoretical analysis.

References

- [1] Green MA. Recent developments in photovoltaics. *Sol Energy* 2004;76:3–8.
- [2] Waldes LC. Competitive solar heat engines. *Renew Energy* 2004;29(11):1825–42.
- [3] Tsoutsos T, Gekas V, Marketaki V. Technical and economical evaluation of solar thermal power generation. *Renew Energy* 2003;28(6):873–86.
- [4] Walker G. *Stirling engines*. Oxford: Clarendon Press; 1980.
- [5] Rogdakis ED. A thermodynamic study for the optimization of stable operation of free piston Stirling engines. *Energ Convers Manage* 2004;45:575–93.
- [6] Bejan JR, Diver RB. The CPG 5-kWe dish-Stirling development program. In: *Proceedings of the IECEC*. San Diego:1992. Paper No. 929181.
- [7] Kongtragool B. A review of solar-powered Stirling engines and low temperature differential Stirling engines. *Renew Sustain Energ Rev* 2003;7:131–54.
- [8] Kongtragool B. Thermodynamic analysis of a Stirling engine including dead volumes of hot space, cold space and regenerator. *Renew Energy* 2006;31:345–59.
- [9] Abdullah S. Design consideration of low temperature differential double acting Stirling engine for solar application. *Renew Energy* 2005;30:1923–41.
- [10] Karabulut H, Yücesu HS, Çınar C. Nodal analysis of a Stirling engine with concentric piston and displacer. *Renew Energy* 2006;31:2188–97.
- [11] Senft JR. Optimum Stirling engine geometry. *Int J Energ Res* 2002;26(12):1087–101.
- [12] Çınar C, Karabulut H. Manufacturing and testing of a gamma type Stirling engine. *Renew Energy* 2005;30(1):57–66.
- [13] Kongtragool B, Wongwises S. Performance of low-temperature differential Stirling engines. *Renew Energy* 2007;32:547–66.